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Application of nanofluids in heat exchangers: A review

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ABSTRACT

The purpose of this review summarizes the important published articles on the enhancement of the convection heat transfer in heat exchangers using nanofluids on two topics. The first section focuses on presenting the theoretical and experimental results for the effective thermal conductivity, viscosity and the Nusselt number reported by several authors. The second section concentrates on application of nanofluids in various types of heat exchangers: plate heat exchangers, shell and tube heat exchangers, compact heat exchangers and double pipe heat exchangers.

Contents

1. Introduction

Heat exchangers are widely used in many engineering applications, for example, applications in chemical industry, power production, food industry, environment engineering, waste heat recovery, air conditioning, and refrigeration.

Nowadays high prices of energy motivate industry to apply energy saving methods as much as possible in their facilities. For decades, efforts have been made to enhance heat transfer of heat exchangers, reduce the heat transfer time and finally improve energy utilization efficiency. These efforts commonly include passive and active methods such as creating turbulence, extending the exchange surface or the use a fluid with higher thermophysical properties.

Recent advances in nanotechnology have allowed development of a new category of liquids termed *nanofluids*, which was first used by Choi [1] to describe liquid suspensions containing nanometer-size particles, including chemically stable metals (e.g., copper, gold, silver), metal oxides (e.g., alumina, bismuth oxide, silica, titania, zirconia), several allotropes of carbon (e.g., diamond, single walled and multi-walled carbon nanotubes, fullerence) with thermal conductivities, orders of magnitudes higher than the base liquids, and with sizes significantly smaller than 100 nm.

The increase in the number of research articles dedicated to this subject thus far shows a noticeable growth and the importance of heat transfer enhancement technology. Thus, this article presents the recent research in convective heat transfer of nanofluids, including the experimental and numerical investigations.

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Nomenclature

Cp	specific heat (J/kg/K)
D	diameter (m)
dp	nanoparticles diameter (m)
f	modeling function
Gr	Grashof number
k	thermal conductivity (W/m/K)
κ_B	Stefan–Boltzmann constant, 1.381×10^{-23} (J/K)
n	empirical shape factor
Nu	Nusselt number
Pe	Peclet number
Pr	Prandtl number
Ra	Rayleigh number
Re	Reynolds number
T	temperature (K)
T_o	reference temperature, 273 K
u_m	average velocity (m/s)

Greek symbols

α	thermal diffusivity coefficient (m^2/s)
β	thermal expansion coefficient (1/T)
δ_v^+	thickness of laminar sublayer (m)
ρ	density (kg/m^3)
ϕ	volume concentration of nanoparticles
μ	dynamic viscosity (N/m/s)

Subscripts

BF	base fluid
eff	effective
f	fluid
w	wall
b	bulk
NP	nanoparticle
NF	nanofluid
p	solid particle

2. Thermophysical properties of nanofluids and dimensionless numbers

The effectiveness of heat transfer is described by the convective heat transfer coefficient, which is a function of a number of thermophysical properties of the nanofluid, the most significant ones being thermal conductivity, specific heat, viscosity and density.

2.1. Thermophysical properties of nanofluids

Thermophysical properties of the nanofluids can be computed using classical formulas derived for a two-phase mixture.

The effective density of the nanofluid is [2]

$$\rho_{NF}=(1-\phi)\rho_{BF}+\phi\rho_{NP}, \tag{1}$$

Specific heat of the nanofluid is calculated from [3] as follows:

$$C_{pNF}=\frac{(1-\phi)(\rho C_p)_{BF}+\phi(\rho C_p)_{NP}}{(1-\phi)\rho_{BF}+\phi\rho_{NP}}, \tag{2}$$

The density and specific heat of the nanofluids are assumed to be a linear function of volume fraction due to lack of experimental data on their temperature dependence.

Thermal conductivity and dynamic viscosity of the nanofluids are dependent not only the volume concentration of nanoparticle, but other parameters such as particle shape (spherical, disk shape or cylindrical), size, mixture combinations and slip mechanisms, surfactant, etc. Experimental studies [4–15] showed that the thermal conductivity as well as viscosity both increases with the addition of nanoparticles.

A summary of selective models on thermal conductivity and dynamic viscosity of nanofluids are listed in Tables 1 and 2.

2.2. Dimensionless numbers

Several theoretical and experimental studies on nanofluid single-phase heat transfer have been reported in literature. The correlations based on the experimental data for finding the Nusselt number of nanofluids from laminar to turbulent flows reported in literature are presented in Table 3.

The dimensionless numbers for the correlations from Table 3 can be computed using following equations:

Reynolds number:

$$Re=\frac{\rho u_m D}{\mu} \tag{3}$$

Prandtl number:

$$Pr=\frac{C_p\mu}{k} \tag{4}$$

Grashof number:

$$Gr=\frac{\beta g D^3\rho^2(T_S-T_\infty)}{\mu^2} \tag{5}$$

Rayleigh number:

$$Ra=Pr\times Gr \tag{6}$$

Peclet number

$$Pe=\frac{u_m d_p}{\alpha} \tag{7}$$

where the thermal diffusivity of nanofluid is

$$\alpha=\frac{k}{(\rho C_p)} \tag{8}$$

Fig. 1 shows the comparison of various proposed correlations for turbulent flow of γ-Al_2O_3/water nanofluid. From Fig. 1 it can be see that the experimental based correlations yields vary close results compared to numerical based correlations due to

Table 1
Summary of the studies on the theoretical and experimental models for effective thermal conductivity of nanofluids.

Model	Reference	Year	Correlation	Relevant information
Theoretical	Maxwell [16]	1881	$\frac{k_{eff}}{k_f}=\frac{k_p+2k_f+2\phi(k_p-k_f)}{k_p+2k_f-\phi(k_p-k_f)}$	Liquid and solid suspensions Spherical particle
	Bruggemann [17]	1935	$\frac{k_{eff}}{k_f}=\frac{1}{4}\left[(3\phi-1)\frac{k_p}{k_f}+(2-3\phi)\right]+\frac{k_f}{4}\sqrt{\Delta}$ $\Delta=\left[\begin{matrix}(3\phi-1)^2\left(\frac{k_p}{k_f}\right)^2+(2-3\phi)^2\\+2(2+9\phi-9\phi^2)\frac{k_p}{k_f}\end{matrix}\right]$	Spherical particles Applicable to high concentrations
	Hamilton and Crosser [18]	1962	$\frac{k_{eff}}{k_f}=\frac{k_p+(n-1)k_f+(n-1)\varphi(k_p-k_f)}{k_p+(n-1)k_f-\varphi(k_p-k_f)}=$ $=4.97\varphi^2+2.72\varphi+1$	$k_p/k_f>100$ Spherical and non-spherical particles Micro-dimensions Various particle shapes
	Wasp [19]	1977	$\frac{k_{eff}}{k_f}=\frac{k_p+2k_f+2\phi(k_p-k_f)}{k_p+2k_f-\phi(k_p-k_f)}$	Hamilton and Crosser's model with $n=3$
	Davis [20]	1986	$\frac{k_{eff}}{k_f}=1+\frac{3(k-1)}{(k-2)-\phi(k-1)}\left[\phi+f(k)\phi^2+O\phi^3\right]$	$f(k)=2.5$ for $k=10$ $f(k)=0.5$ for $k=\infty$
	Lu and Lin [21]	1996	$\frac{k_{eff}}{k_f}=1+a\phi+b\phi^2$	Spherical and non-spherical particles For $k=10$: $a=2.25$, $b=2.27$ For $k=\infty$: $a=3.00$, $b=4.51$
	Bhattacharya et al. [22]	2004	$\frac{k_{eff}}{k_f}=\frac{k_p}{k_f}\phi+(1-\phi)$ $k_p=\frac{1}{\kappa_B T^2 V}\sum_{j=0}^{n}(Q(0)Q(j\Delta T))\Delta T$	Brownian dynamics
	Koo and Kleinstreuer [23], [24]	2004 2005	$\frac{k_{eff}}{k_f}=\frac{k_p+2k_f+2\phi(k_p-k_f)}{k_p+2k_f-\phi(k_p-k_f)}+5\times 10^4\beta\rho_p c_p$ $\sqrt{\frac{\kappa_B T}{\rho_p d_p}}f(T,\phi)$ $f(T,\phi)=(-134.63+1722.3\phi)+(0.4705-6.04\phi)\frac{T}{T_0}$ $\beta=\begin{cases}0.0137(100\phi)^{-0.8229}\phi<0.01\\0.0011(100\phi)^{-0.7272}\phi>0.01\end{cases}$	CuO/ethylene glycol CuO/oil Considered surrounding liquid traveling with randomly moving nanoparticles
	Prasher et al. [25]	2005	$\frac{k_{eff}}{k_f}=(1+ARe^m Pr^{0.333}\phi)\left(\frac{k_p+2k_f+2\phi(k_p-k_f)}{k_p+2k_f-\phi(k_p-k_f)}\right)$	Effect of convection of the liquid near the particle included A and m are constants Nanospheres
	Xue [26]	2005	$\frac{k_{eff}}{k_f}=\frac{1-\phi+2\phi\frac{k_p}{k_p-k_f}\quad \ln\frac{k_p+k_f}{2k_f}}{1-\phi+2\phi\frac{k_f}{k_p-k_f}\quad \ln\frac{k_p+k_f}{2k_f}}$	Nanospheres with interfacial shell
Experimental	Li and Peterson [27]	2006	$\frac{k_{eff}-k_f}{k_f}=0.764\phi+0.0187(T-273.15)-0.462$ $\frac{k_{eff}-k_f}{k_f}=3.761\phi+0.0179(T-273.15)-0.307$	Al_2O_3/water nanofluids CuO/water nanofluids
	Buongiorno [28]	2006	$\frac{k_{eff}}{k_f}=1+2.92\phi-11.99\phi$	TiO_2/water nanofluids
	Timofeeva et al.[29]	2007	$k_{NF}=(1+3\phi)k_f$	Al_2O_3/water nanofluids
	Avsec and Oblak [30]	2007	$\frac{k_{eff}}{k_f}=\left[\frac{k_p+(n-1)k_f+(n-1)(1+\beta)^3\phi(k_p-k_f)}{k_p+(n-1)k_f-(1-\beta)^3\phi(k_p-k_f)}\right]$	$n=(3/\psi)$- empirical shape factor Al_2O_3/ethylene glycol Cu/ethylene glycolTiO_2/water
	Chandrsekar et al. [31]	2009	$\frac{k_{eff}}{k_f}=\left[\frac{k_p+(n-1)k_f+(n-1)(1+\beta)^3\phi(k_p-k_f)}{k_p+(n-1)k_f-(1-\beta)^3\phi(k_p-k_f)}\right]+\frac{C\phi(T-T_0)}{\mu k a^4}$	Al_2O_3/water CuO/water TiO_2/water TiO_2/ethylene glycol
	Duangthongsuand Wongwises [32]	2009	$\frac{k_{eff}}{k_f}=a+b\phi$ $a=1.0225$, $b=0.0272$ for $T=15\ ^\circ C$ $a=1.0204$, $b=0.0249$ for $T=25\ ^\circ C$ $a=1.0139$, $b=0.0250$ for $T=35\ ^\circ C$	TiO_2/water nanofluids
	Patel et al. [33]	2010	$\frac{k_{eff}}{k_f}=\left(1+0.135\left(\frac{k_p}{k_f}\right)^{0.273}\phi^{0.467}\left(\frac{T}{20}\right)^{0.547}\left(\frac{100}{d_p}\right)^{0.234}\right)$	Oxide and metallic nanofluids
	Chandrasekar et al. [34]	2010	$\frac{k_{eff}}{k_f}=\left(\frac{C_{p,eff}}{C_{p,f}}\right)^a\left(\frac{\rho_{eff}}{\rho_f}\right)^b\left(\frac{M_f}{M_{eff}}\right)^c a=-0.023,\quad b=1.358,\quad c=0.125$	Al_2O_3/water nanofluids
	Vajjha et al. [35]	2010	$\frac{k_{eff}}{k_f}=\frac{k_p+2k_f+2\phi(k_p-k_f)}{k_p+2k_f-\phi(k_p-k_f)}+5\times 10^4\beta\rho_p c_p$ $\sqrt{\frac{\kappa_b T}{\rho_p D}}f(T,\phi)$ $f(T,\phi)=(-3.0669\cdot 10^{-2}\phi-3.91123\times 10^{-3})+(2.8217\times 10^{-2}\phi+3.917\times 10^{-3})\frac{T}{T_0}$	Al_2O_3/(60:40) EG/water nanofluids

Table 1 (*continued*)

		$\beta = \begin{cases} 8.4407(100\phi)^{-1.07304} 0.01 \langle \phi \rangle 0.1 \text{ for } Al_2O_3 \\ 9.881(100\phi)^{-9446} 0.01 \langle \phi \rangle 0.06 \text{ for CuO} \end{cases}$	CuO/(60:40)
			EG/water nanofluids
Godson et al. [36]	2010	$\frac{k_{eff}}{k_f} = 0.9692\phi + 0.9508$	Ag/water nanofluids
Corcione [37]	2011	$\frac{k_{eff}}{k_f} = 1 + 4.4Re^{0.4}Pr^{0.66}\left(\frac{T}{T_{fr}}\right)^{10}\left(\frac{k_p}{k_f}\right)^{0.03}\phi^{0.66}$	Al_2O_3/water nanofluids

Table 2
Summary of the studies on the theoretical and experimental models for effective viscosity of nanofluids.

Model	Reference	Year	Correlation	Relevant information
Theoretical	Einstein [38]	1906	$\frac{\mu_{eff}}{\mu_f} = 1 + 2.5\phi$	Infinitely dilute suspension of spheres
	Saito [39]	1950	$\frac{\mu_{eff}}{\mu_f} = \left(1 + \frac{2.5}{(1-\varphi)}\varphi\right)$	Spherical rigid particles
				Brownian motion Very small particles
	Brinkman [40]	1952	$\frac{\mu_{eff}}{\mu_f} = \frac{1}{(1-\phi)^{2.5}}$	Spherical particles
				Valid for high moderate particle concentrations
	Lundgren [41]	1972	$\frac{\mu_{eff}}{\mu_f} = \frac{1}{1-2.5\phi}$	Dilute concentration of spheres
	Batchelor [42]	1977	$\frac{\mu_{eff}}{\mu_f} = 1 + 2.5\phi + 6.2\phi^2$	Rigid and spherical particles
				Brownian motion Isotropic structure
Experimental	Drew and Passman [43]	1999	$\frac{\mu_{NF}}{\mu_f} = 1 + 2.5\phi$	$\phi < 5.0$ vol%
				Cu/water, Au, CNT, grapheme
	Wang et al. [44]	1999	$\frac{\mu_{eff}}{\mu_f} = 1 + 7.3\varphi + 123\varphi^2$	Al_2O_3/water
				Al_2O_3/ethylene glycol
	Tseng and Lin [45]	2003	$\frac{\mu_{eff}}{\mu_f} = 13.47\exp(35.98\phi)$	TiO_2/water nanofluids
	Maiga et al. [46]	2005	$\frac{\mu_{eff}}{\mu_f} = 1 + 7.3\varphi + 123\varphi^2$	Al_2O_3/water nanofluids
	Maiga et al. [46]	2005	$\frac{\mu_{eff}}{\mu_f} = 1 - 0.19\phi + 306\phi^2$	Al_2O_3/ethylene glycol nanofluids
	Song et al. [47]	2005	$\frac{\mu_{eff}}{\mu_f} = 1 + 56.5\varphi$	SiO_2/water nanofluids
	Koo and Kleinstreuer [48]	2005	$\mu_{Brownian} = 5 \times 10^4 \beta \rho_f \phi$ $\sqrt{\frac{\kappa_b T}{2\rho_p r_p}\left[(-134.63 + 1722.3\phi) + (0.4705 - 6.04\phi)\frac{T}{T_0}\right]}$ $\beta = \begin{cases} 0.0137(100\phi)^{-0.8229} \phi < 0.01 \\ 0.0011(100\phi)^{-0.7272} \phi > 0.01 \end{cases}$	CuO/water nanofluids
	Kulkarni et al. [49]	2006	$\ln\mu_{eff} = -(2.8751 + 53.548\phi - 107.12\phi^2) + (1078.3 + 15857\phi + 20587\phi^2)\left(\frac{1}{T}\right)$	CuO/water nanofluids
	Buongiorno [28]	2006	$\frac{\mu_{eff}}{\mu_f} = 1 + 5.45\phi + 108.2\phi^2$	TiO_2/water nanofluids
			$\frac{\mu_{eff}}{\mu_f} = 1 + 39.11\phi + 533.9\phi^2$	Al_2O_3/water nanofluids
	Chen et al. [50]	2007	$\frac{\mu_{eff}}{\mu_f} = 1 + 10.6\phi + 112.36\phi^2$	TiO_2/ethylene glycol nanofluids
	Nguyen et al. [51]	2007	$\frac{\mu_{eff}}{\mu_f} = 0.904 \quad \exp(0.1483\ \phi) \quad dp = 47$ nm	Al_2O_3/water nanofluids
			$\frac{\mu_{eff}}{\mu_f} = 1 + 0.025\phi + 0.015\phi^2 \quad dp = 36$ nm	
			$\frac{\mu_{eff}}{\mu_f} = 1.475 - 0.319\phi + 0.051\phi^2 + 0.009\phi^3$ dp = 29 nm	CuO/water nanofluids
	Namburu et al. [52]	2007	$\mathrm{Log}(\mu_{eff}) = Ae^{-BT}$ $A = 1.8375\phi^2 - 29.643\phi + 165.56$ $B = 4 \times 10^{-6}\phi^2 - 0.001\phi + 0.0186$	CuO/(60:40)
				EG/water nanofluids
	Grag et al. [53]	2008	$\frac{\mu_{eff}}{\mu_f} = 1 + 11\phi$	Cu/ethylene glycol nanofluids
	Masoumi et al. [54]	2009	$\mu_{eff} = \mu_f + \frac{\rho_p V_B d_p^2}{72C\delta}$ $\delta = \sqrt[3]{\frac{\pi}{6\phi}}d_p$	Al_2O_3/water nanofluids
	Duangthongsuand Wongwises [32]	2009	$\frac{\mu_{eff}}{\mu_f} = a + b\phi + c\phi^2$ $a = 1.0226$, $b = 0.0477$, $c = -0.0112$ for $T = 15$ °C	TiO_2/water nanofluids

Table 2 (*continued*)

Model	Reference	Year	Correlation	Relevant information
			$a=1.0130$, $b=0.0920$, $c=-0.0150$ for $T=25$ °C $a=1.0180$, $b=0.1120$, $c=-0.0177$ for $T=35$ °C	
	Chandrasekar et al. [34]	2010	$\frac{\mu_{eff}}{\mu_f}=1+b\left(\frac{\phi}{1-\phi}\right)^n$ $b=1631, \quad n=2.8$	Al_2O_3/water nanofluids
	Vajjha [35]	2010	$\frac{\mu_{eff}}{\mu_f}=Ae^{C\phi}$ $A=0.9197, \quad C=22.8539$ $\mu_b=Ae^{B/T}$ $A=0.555\times10^{-3}, \quad B=2664$	CuO/(60:40)
	Corcione [37]	2011	$\frac{\mu_{eff}}{\mu_f}=\frac{1}{1-34.87(d_p/d_f)^{-0.3}\phi^{-1.03}}$ $d_f=0.1+\left(\frac{6M}{N\pi\rho_{f.0}}\right)^{1/3}$	EG/water nanofluids SiO_2/ethanol nanofluids

Table 3
Convective heat transfer correlations.

Reference	Year	Correlation	Relevant information
Pak and Cho [55]	1998	$Nu=0.021Re^{0.8}Pr^{0.5}$	Experimental study Turbulent flow Al_2O_3/water nanofluids TiO_2/water nanofluids $0<\phi<3.0$ vol% $10^4<Re<10^5$ $6.5<Pr<12.3$
Li and Xuan [56]	2002	$Nu=0.4328(1+11.285\phi^{0.754}Pe^{0.218})Re^{0.333}Pr^{0.4}$	Experimental study Laminar flow Cu/water nanofluids $0<\phi<2.0$ vol% $Re>800$
Xuan and Li [57]	2003	$Nu=0.0059(1+7.6286\phi^{0.6886}Pe^{0.001})Re^{0.9238}Pr^{0.4}$	Experimental study Turbulent flow Cu/water nanofluids $0<\phi<2.0$ vol% $10^4<Re<2.5\times10^4$
Yang et al. [58]	2005	$Nu=a\mathrm{Re}^b\mathrm{Pr}^{1/3}\left(\frac{D}{L}\right)^{1/3}\left(\frac{\mu_w}{\mu_b}\right)^{-0.14}$	Experimental study Laminar flow Graphite-synthetic oil nanofluid $0<\phi<2.0$ vol% $5<Re<110$
Maiga [46]	2005	$Nu=0.28\mathrm{Re}^{0.35}\mathrm{Pr}^{0.35}$ for constant temperature $Nu=0.086\mathrm{Re}^{0.55}\mathrm{Pr}^{0.5}$ for constant wall heat flux	Numerical study Laminar flow Al_2O_3/water nanofluids $0<\phi<10.0$ vol% $Re\leq1000$ $6.0<Pr<753$
Maiga [59]	2006	$Nu=0.085\mathrm{Re}^{0.71}\mathrm{Pr}^{0.35}$	Numerical study Turbulent flow Al_2O_3/water nanofluids $0<\phi<10.0$ vol% $104<Re<5\times105$ $6.6<Pr<13.9$
Buongiorno [28]	2006	$Nu=\frac{(f/8)(Re-1000)Pr}{1+\delta_v^+\sqrt{(f/8)}(Pr_v^{2/3}-1)}$	Numerical study Fully developed turbulent flow
Duangthongsuk and Wongwises [60]	2010	$Nu=0.074\mathrm{Re}^{0.707}\mathrm{Pr}^{0.385}\varphi^{0.074}$	Experimental study Turbulent flow TiO_2/water nanofluids $0.2<\phi<2.0$ vol% $3000<Re<1.8\times10^4$
Vajjha et al. [61]	2010	$Nu=0.065(Re^{0.65}-60.22)(1+0.0169\phi^{0.15})Pr^{0.542}$	Experimental study Turbulent flow Al_2O_3/water nanofluids CuO/water nanofluids SiO_2/water nanofluids $0<\phi<6.0$ vol% for CuO, SiO_2 $0<\phi<10.0$ vol% for $Al_2O_3 3000<Re<1.6\times10^4$

Table 3 (*continued*)

Sajadi and Kazemi [62]	2011	$Nu = 0.067\mathrm{Re}^{0.71}\mathrm{Pr}^{0.35} + 0.0005\mathrm{Re}$	Experimental study Turbulent flow TiO_2/water nanofluids $0.2 < \phi < 0.25$ vol% $5000 < Re < 3 \times 10^4$
Godson Asirvatham et al. [63]	2011	$Nu = 0.023Re^{0.8}Pr^{0.3} + (0.617\phi - 0.135)Re^{(0.445\phi - 0.37)}Pr^{(1.081\phi - 1.305)}$	Experimental study
Mansour et al. [64]	2011	Horizontal tube: $Nu = Nu_0(1-\phi^{0.625})(1+5.25 \times 10^{-5} \times Ra^{1.06})^{0.135}$ Vertical tube: $Nu = Nu_0\left(1+52 \times 10^{-4} \times \frac{Gr}{Re}\right)^{0.28}$	Experimental study Laminar flow Al_2O_3/water nanofluids $0 < \phi < 4.0$ vol% $350 < Re < 900$ $5 \times 10^5 < Ra < 9.6 \times 10^5$

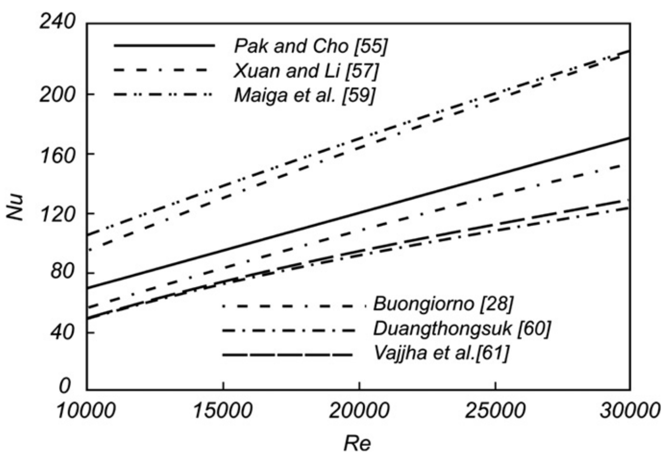

Fig. 1. Comparison of proposed correlations for Nusselt number versus Reynolds number of γ-Al_2O_3/water nanofluid under turbulent flow (Sarkar [84]).

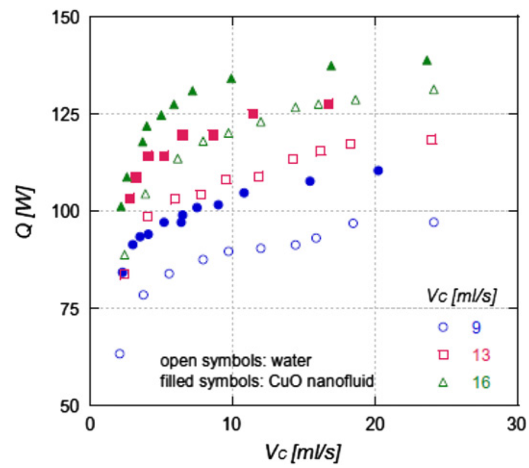

Fig. 2. Heat flow rate vs. cooling liquid flow rate for various hot water flow rates (Pantzalia et al. [88]).

misleading assumptions of physical mechanisms of nanofluids in numerical study.

3. Application of nanofluids

Convective heat transfer is one of the most widely investigated thermal phenomena in nanofluids [55–77], relevant to a number of engineering applications. Due to the observed improvement in the thermal conductivity, nanofluids are expected to provide enhanced convective heat transfer coefficients. However, as the suspensions of nanoparticles in the base fluids affect the thermophysical properties other than thermal conductivity also, such as the viscosity and the thermal capacity, quantification of the influence of nanoparticles on the heat transfer performance is essentially required [78].

Godson et al. [78], Wang and Mujumdar [79], Duangthongsuk and Wongwises [80], Kakaç and Pramuanjaroenkij [81], Wen et al. [82], Mohammed et al. [83], Sarkar [84], Murshed et al. [85], Saidur et al. [86] and Vajjha and Das [87] presented reviews of nanofluids for heat transfer applications. From the observed results it is clearly seen, that nanofluids have greater potential for heat transfer enhancement and are highly suited to application in different types of heat exchangers.

This section explains application of nanofluids in heat exchangers used in industries, such as plate heat exchangers, shell-and-tube heat exchangers, compact heat exchangers and double-pipe heat exchangers, based on available literatures.

3.1. Plate heat exchangers

Pantzali et al. [88] studied numerically and experimentally the effects of nanofluids on the performance of a miniature plate heat exchanger with modulated surface. Their thermophysical measurements of the nanofluid (CuO in water, 4 vol%) reveal that the increase in thermal conductivity is accompanied by a significant drop in heat capacity and an increase in viscosity. Besides that, it was outlined that the heat transfer enhancement is more at lower flow rate and at higher flow rate, where the main heat transport mechanism is convection, nanoparticle contribution is limited (Fig. 2). The results suggest that, for a given heat duty, the nanofluid volumetric flow rate required is lower than that of water causing lower pressure drop, resulting in lower pressure drop and less pumping power.

Also, Pantzali et al. [89] performed a study on plate heat exchangers (PHE) using nanofluids (CuO nanoparticles and water) as coolants. Their methodology included preparation of the nanofluids, measurement of the thermophysical properties of the nanofluids, experimental analysis and also CFD simulation to gain an insight of the flow inside the PHE. The experimental apparatus of their work was depicted in Fig. 3. The PHE comprises 16 stainless steel corrugated plates that create two isolated fluid paths for the hot and cold fluid flow, respectively, forming 8 flow channels per stream. The plates have chevron-type corrugations with a height of 2 mm and a wave length (in a direction normal to the crests) 8.6 mm. The corrugations form a herringbone pattern with an angle of 50° relative to the direction of the flow. Successive plates are arranged with the corrugation pattern pointing in opposite directions. Water is used as the hot fluid and its temperature is controlled at approximately 50 °C

by means of a heater, while its flow rate is adjusted using a high-accuracy valve and is measured by a float-type flow meter. Either water or the nanofluid is used as cooling liquid in the PHE with the intention of comparing their performance. The cooling liquid is stored in a 5-l container and is recirculated by a centrifugal pump. Its inlet temperature is controlled at approximately 30 °C using another similar PHE with tap water as the working fluid.

The results from the measurement of the thermophysical properties of the nanofluids yielded these findings: increase of thermal conductivity; increase of density; decrease of heat capacity; increase in viscosity; and possible non-Newtonian behavior. The experimental data showed that the thermo physical properties and type of flow inside the heat exchanger played important roles in the efficiency of the nanofluid as a coolant. It was concluded that in industrial heat exchangers where large volumes of nanofluids are required and the flow is turbulent, the use of nanofluids seems impractical.

Maré et al. [90] investigated the thermal performances of two types of nanofluids (oxides of alumina dispersed in water and aqueous suspensions of nanotubes of carbons) in two plate heat exchangers. The experimental system used in this study was shown schematically in Fig. 4.

The experimental apparatus mainly consisted of three buckles (hot buckle, central buckle, cold buckle) and two identic plate heat exchangers. The central buckle is equipped with an UPS circulator 60 Grundfos with three speeds assuring a flow rate between 400 l/h and 1000 l/h, of an ultrasonic flow meter and of four PT100 probes permitting the measure of four temperatures (T_{ce}, T_{cs}, T_{fe}, T_{fs}). Distilled water in the case of the validation of the installation as well as the different nanofluids will circulate in this buckle. The production of heat is assured by a vulcatherm

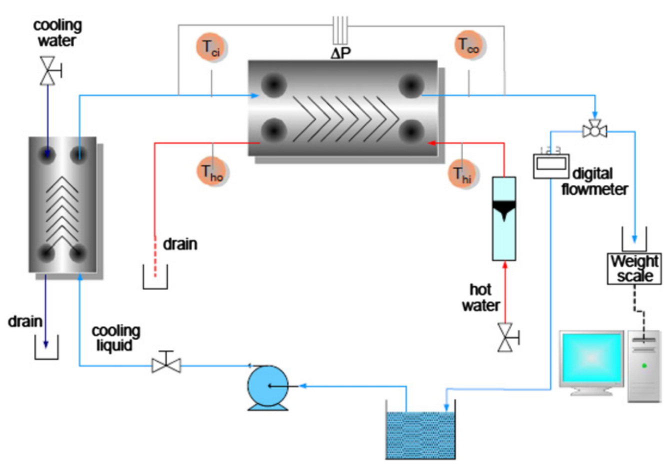

Fig. 3. Experimental setup for heat transfer study in plate heat exchangers using CuO–water nanofluids as coolants (Pantzali et al. [89]).

vulcanix of a calorific power of 2 kW with a range of temperature comprised between 10 °C and 90 °C, in the hot buckle (*V* indication). Two PT100 probes permit the measures of the entry and exit temperatures of the exchanger (T_{ve}, T_{vs}). The coolant fluid is water and a system of regulation permits to control the entry temperature T_{ve} intersection (hot side). The volumic flow rate (Q_{v1}) is measured by an electromagnetic flow meter.

The results showed a significant enhancement in laminar mode of the convective heat transfer coefficient of about 42% and 50% for alumina and carbon nanotubes, respectively compared to that of pure water for the same Reynolds number. Also, the results showed that the impact of the viscosity and the pressure drop at low temperatures is important and has to take into account before to use nanofluids in heat exchanger. Finally, we observed that the gain can reach 22% for alumina and 150% for carbon nanotubes. This result reports that alumina and carbon nanotubes showed a better thermal–hydraulic performance in terms of a competition between heat transfer enhancement and pumping power loss in comparison with pure water.

Experimental investigations on the heat transfer characteristics and pressure drop of the ZnO and Al_2O_3 nanofluids in a plate heat exchanger have been reported by Kwon et al. [91]. The experimental conditions were 100–500 Reynolds number and the respective volumetric flow rates. The working temperature of the heat exchanger was within 20–40 °C. The measured thermophysical properties, such as thermal conductivity and kinematic viscosity, were applied to the calculation of the convective heat transfer coefficient of the plate heat exchanger employing the ZnO and Al_2O_3 nanofluids made through a two-step method. Experimental results showed that, according to the Reynolds number, the overall heat transfer coefficient for 6 vol% Al_2O_3 increased to 30% because at the given viscosity and density of the nanofluids, they did not have the same flow rates. At a given volumetric flow rate, however, the performance did not improve. After the nanofluids were placed in the plate heat exchanger, the experimental results pertaining to nanofluid efficiency seemed inauspicious.

The study of convective heat transfer in a corrugated plate heat exchanger using nanofluid, containing aluminum oxide in water (as base fluid), in different concentrations and water in turbulent flow have been reported by Pandey and Nema [92]. Results showed that the heat transfer characteristics improve with increase in Reynolds- and Peclet-number and with decrease in nanofluid concentration. For a given heat load, the required pumping power increased with increase in nanofluid concentration. Both power consumption and heat transfer rates were lower for water in comparison to the nanofluid for flow rates of 2–5 lpm for hot and cold fluids. Further, for a given heat load the nanofluid required lower flow rate but suffered higher pressure drop than that for water. For a given pumping power more heat could be removed by the nanofluids relative to water, though the maximum heat transfer rate was found with the lowest concentration of nanofluids.

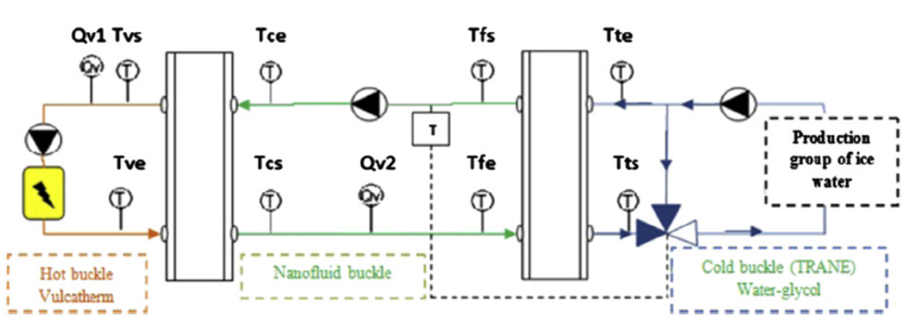

Fig. 4. Experimental setup for the study of thermal performances of γ-Al_2O_3/water and CNTs/water nanofluids in a plate heat exchanger at low temperature (Maré et al. [90]).

3.2. Shell and tube heat exchangers

The characteristics of Al_2O_3/ethylene glycol nanofluid and ethylene glycol fluid which cross a rectangular arrangement of tubes in a shell and tubes heat exchanger have been reported by Khoddamrezaee et al. [93]. The stagnation point, separation point, heat transfer coefficient and shear stress in both of nanofluid and pure fluid have been determined and compared with each other. The results showed that by using of nanofluids, the stagnation and separation points of flow were postponed and the amount of heat transfer coefficient and shear stress increased but the effect of shear stress increase can be neglected in compare of unusual heat transfer rising.

Farajollahi et al. [94] performed an experimental analysis to study of the heat transfer characteristics of nanofluids in a shell and tube heat exchanger. The nanofluids used were γ-Al_2O_3/water and TiO_2/water under turbulent flow conditions to investigate the effects of Peclet number, volume concentration of suspended particles, and particle type on the heat transfer characteristics. The experimental system used in this study was shown schematically in Fig. 5. This mainly includes two flow loops (nanofluids and water flow loops). The test section is a shell and tube heat exchanger where nanofluid passes through the 16 tubes with 6.1 mm outside diameter, 1 mm thickness, and 815 mm length and water flows inside the shell with 55.6 mm inside diameter. The tube pitch is 8 mm and the baffle cut and baffle spacing are 25% and 50.8 mm, respectively. The experiments were performed within the following ranges: the nanoparticle volume concentrations of γ-Al_2O_3/water and TiO_2/water nanofluids vary in the range of 0.3–2% and 0.15–0.75%, respectively, and the Peclet number varied between 20,000 and 60,000.

The results have indicated that addition of nanoparticles to the base fluid enhances the heat transfer performance and results in larger heat transfer coefficient than that of the base fluid at the same Peclet number. It was noticed that heat transfer characteristics of nanofluids increase significantly with Peclet number. TiO_2/water and γ-Al_2O_3/water nanofluids possess better heat transfer behavior at the lower and higher volume concentrations, respectively. The experimental results were also in agreement with the predicted values of available correlation at the lower nanoparticle volume concentrations.

Lotfi et al. [95] conducted a experimentally study on heat transfer enhancement of multi-walled carbon nanotube (MWNT)/water nanofluid) in a horizontal shell and tube heat exchanger. The test section is the heat exchanger, in which nanofluid passes through 14 tubes with 7 mm inside diameter and 580 mm length, while the coolant flows through the shell-side with a 101 mm inside diameter. The inlet and outlet parts of heat exchanger were equipped with four K type thermocouples. Measurement error in K type thermocouples for determining fluid temperatures was ± 0.1 °C. The heating section consists of a horizontal copper tube with 11.42 mm inner diameter and 100 mm length. The tube surface is electrically heated by the use of an AC power supply for generating. In this work carbon nanotubes were synthesized by the use of catalytic chemical vapor deposition (CCVD) method over Co–Mo/MgO nanocatalyst. Obtained MWNTs were purified using a three stage method. COOH functional groups were inserted for making the nanotubes hydrophilic and increasing the stability of the nanofluid. The results indicate that the presence of multi-walled nanotubes enhances the heat transfer rate in a shell and tube heat exchanger (Fig. 6).

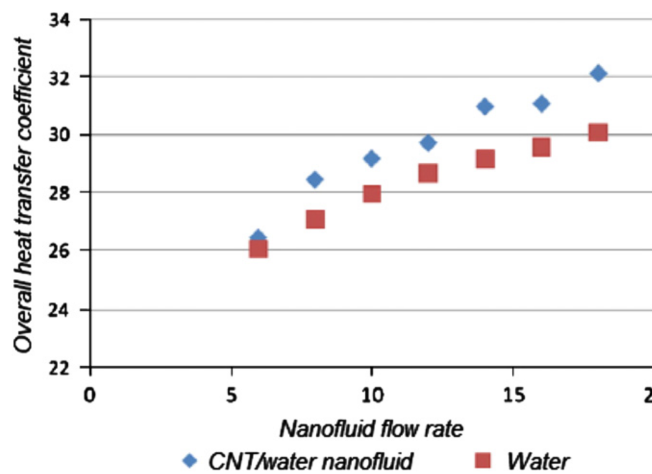

Fig. 6. Comparison between the measured overall heat transfer coefficient for water and nanofluid for Q=630 W (Lotfi et al. [95]).

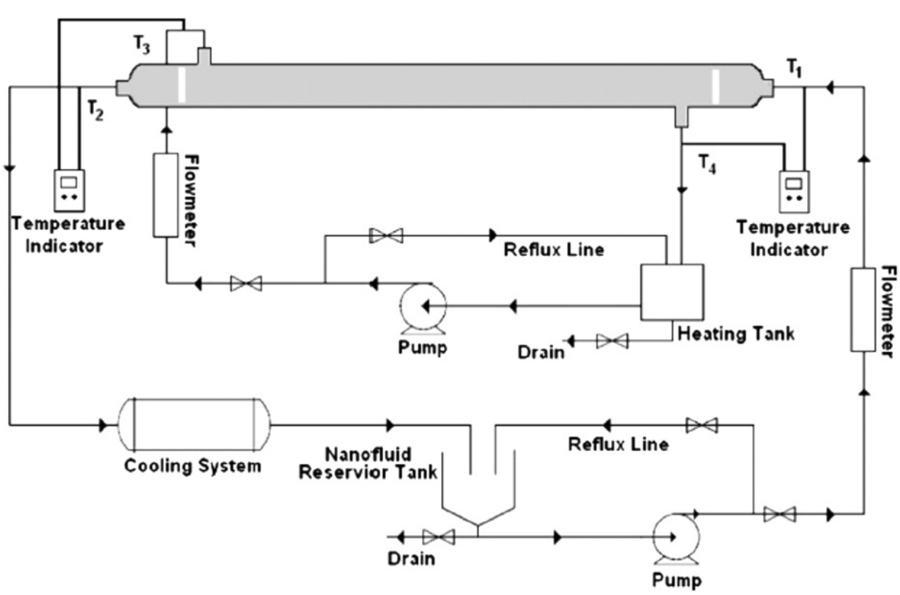

Fig. 5. Experimental setup for the study of the heat transfer characteristics of γ-Al_2O_3/water and TiO_2/water nanofluids in a shell and tube heat exchanger under turbulent flow conditions (Farajollahi et al. [94]).

Leong et al. [96] investigated the application of nanofluids as working fluids in shell and tube heat recovery exchangers in a biomass heating plant. Heat exchanger specification, nanofluid properties and mathematical formulations were taken from the literature to analyze thermal and energy performance of the heat recovery system. The results showed that the convective and overall heat transfer coefficient increased with the application of nanofluids compared to ethylene glycol or water based fluids. It addition, 7.8% of the heat transfer enhancement could be achieved with the addition of 1% copper nanoparticles in ethylene glycol based fluid at a mass flow rate of 26.3 kg/s and 116.0 kg/s for flue gas and coolant, respectively.

3.3. Compact heat exchangers

Compact heat exchanger is a unique and special class of heat exchanger having a large heat transfer area per unit volume. These have been widely used in various applications in thermal fluid systems including automotive thermal fluid systems In addition, flat tubes are more popular in automotive applications due to the lower drag profile compared to round tubes.

Vasu et al. [97] studied the thermal design of flat tube plain fin compact heat exchanger with the ε-NTU rating method using Al_2O_3/water nanofluid as coolant. The results showed that the pressure drop of 4% nanoparticles of Al_2O_3 is almost double of the base fluid (Fig. 7).

Vajjha et al. [35] numerically investigated the cooling performance of a flat tube of a radiator under laminar flow with two different nanofluids, Al_2O_3 and CuO, in ethylene glycol and water mixture. Numerical results showed that at a Reynolds number of 2000, the percentage increase in the average heat transfer coefficient over the base fluid for a 10% Al_2O_3 nanofluid is 94% and that for a 6% CuO nanofluid is 89%. Also, the analysis shows that the average heat transfer coefficient increases with the Reynolds number and the particle volumetric concentration (Fig. 8). For constant inlet velocity, an increase in the particle volume concentration results in an increase in the skin friction coefficient along the duct. The average skin friction coefficient for a 6% CuO nanofluid in the fully developed region is about 2.75 times in comparison to that of the base fluid at a constant inlet velocity of 0.3952 m/s. For the same amount of heat transfer, the

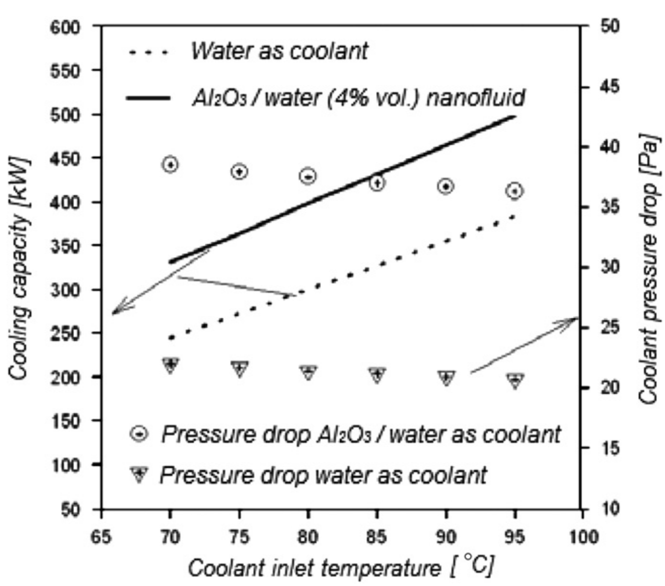

Fig. 7. Coolant inlet temperature influence on the thermal and fluid dynamic performance of compact heat exchanger for M_a=12 kg/s and $T_{air\ inlet}$=20 °C (Vasu et al. [97]).

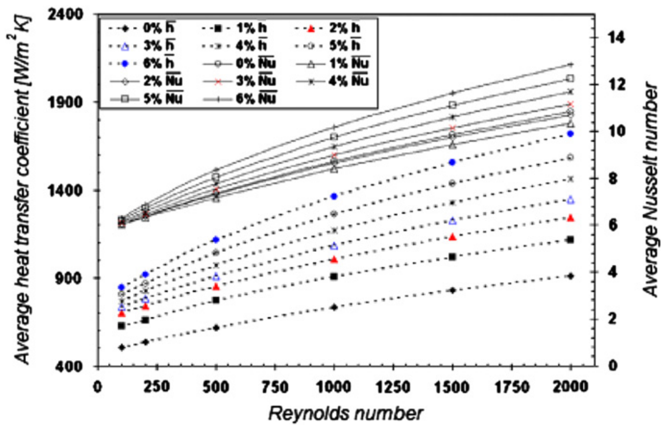

Fig. 8. Variation of h and Nu with Re number for different particle volumetric concentrations of CuO nanofluid (Vajjha et al. [35]).

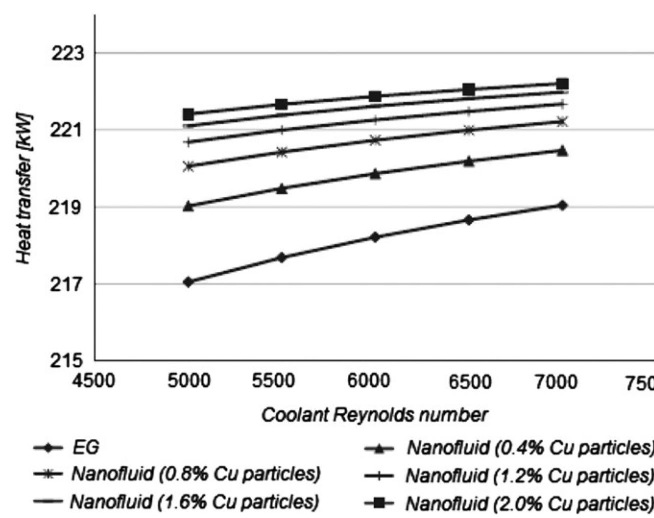

Fig. 9. Effect of coolant Reynolds number to heat transfer rate of radiator (Leong et al. [98]).

pumping power requirement is 82% lower for Al_2O_3 nanofluid of 10% concentration and 77% lower for CuO nanofluid of 6% concentration when compared to the base fluid.

Leong et al. [98] studied the application of ethylene glycol based copper nanofluids in an automotive cooling system. The results showed that, overall heat transfer coefficient and heat transfer rate in engine cooling system increased with the usage of nanofluids (with ethylene glycol the basefluid) compared to ethylene glycol (i.e., basefluid) alone (Fig. 9). Also, it is observed that, about 3.8% of heat transfer enhancement could be achieved with the addition of 2% copper particles in a basefluid at the Reynolds number of 6000 and 5000 for air and coolant, respectively.

Peyghambarzadeh et al. [99] experimentally analyzed forced convective heat transfer of Al_2O_3/water nanofluids in an automobile radiator under turbulent flow. The experimental system (Fig. 10) used in this research includes flowlines, a storage tank, a heater, a centrifugal pump, a flow meter, a forced draft fan and a cross flow heat exchanger (an automobile radiator). The working fluid fills 25% of the storage tank whose total volume is 30 l (height of 35 cm and diameter of 30 cm). The total volume of the circulating liquid is constant in all the experiments. For heating theworking fluid, an electrical heater and a controller were used to maintain the temperature between 40 °C and 80 °C. Five different concentrations of nanofluids in the range of 0.1–1 vol%

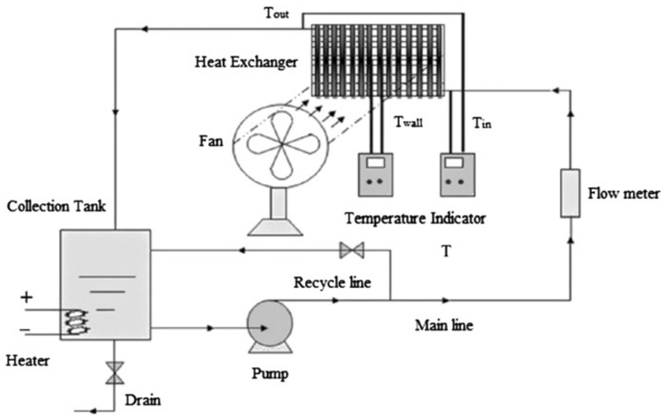

Fig. 10. Experimental setup for the study of the convective heat transfer of Al_2O_3/water nanofluid in an automobile radiator under turbulent flow (Peyghambarzadeh et al. [99]).

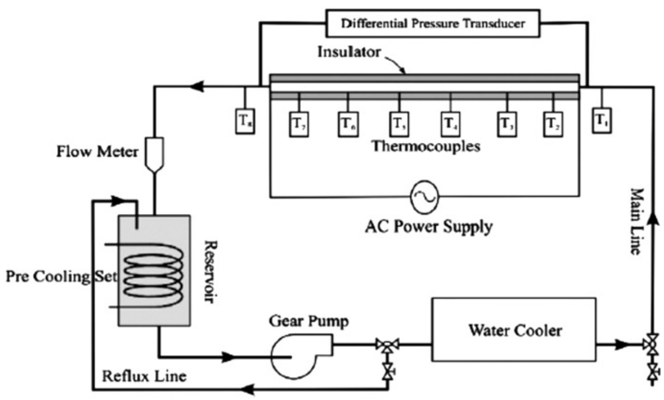

Fig. 11. Experimental setup for the study of the flow and heat transfer characteristics of the CuO-base oil nanofluid flow inside the round tube and flattened tubes under constant heat flux (Razi et al. [101]).

have been prepared by the addition of Al_2O_3 nanoparticles into the water. Experimental results showed that increasing the fluid circulating rate can improve the heat transfer performance while the fluid inlet temperature to the radiator has trivial effects. Meanwhile, application of nanofluid with low concentrations can enhance heat transfer efficiency up to 45% in comparison with pure water.

The convective heat transfer enhancement of Al_2O_3/water and Al_2O_3/ethylene glycol nanofluids as the coolants inside flat aluminum tubes of the car radiator has been investigated by Peyghambarzadeh et al. [100]. Significant increases of total heat transfer rates have been observed with the nanoparticle addition. A highest Nusselt number enhancement up to 40% was obtained at the best conditions for both nanofluids. The experimental results have demonstrated that the heat transfer behaviors of the nanofluids were highly depended on the particle concentration and the flow conditions and weakly dependent on the temperature. Also, the results showed that, the heat transfer enhancement of about 40% compared to the base fluids.

Razi et al. [101] studied heat transfer and pressure drop characteristics of the pure base oil and CuO-base oil nanofluid flow inside the round tube and flattened tubes under constant heat flux. Five round copper tubes of 12.7 mm outer diameter, 0.6 mm wall thickness and 1200 mm length are selected. Four tubes of them are flattened into oblong shapes with internal heights of 9.6 mm, 8.3 mm, 7.5 mm, and 6.3 mm and the fifth one is used as a round tube. The experimental apparatus used in this study was shown schematically in Fig. 11.

To evaluate the overall performance of the two enhanced heat transfer techniques utilized, a new parameter called "performance index" was defined to consider both heat transfer and pressure drop characteristics, simultaneously.

Experimental results showed that, for a given flattened tube and at a same flow conditions, there is a noticeable increase in heat transfer coefficient as well as pressure drop of nanofluids compared to that of base liquid. Also, at the same flow conditions and for a given nanofluid with constant particle concentration, flattened tubes enhance the heat transfer rates compared to that of the round tube, significantly. As the tube profile is more flattened, this enhancement is more pronounced. The same enhancement trend in pressure drop is seen when the tube profile is becoming more flattened. Nanofluids have better heat transfer characteristics when they flow in flattened tubes rather than in the round tube. Compared to pure oil flow, maximum heat transfer enhancement of 16.8%, 20.5% and 26.4% is obtained for nanofluid flow with 2% wt. concentration inside the round tube and flattened tubes with internal heights of 8.3 mm and 6.3 mm, respectively.

The cooling performance of an automobile radiator under laminar flow with nanofluids is numerically investigated by Huminic [102]. Results showed that at a Reynolds number of 10 the cooling performance of their automobile radiator with Cu-ethylene glycol (2 vol%) nanofluid is enhanced by about 8% compared to that of with ethylene glycol alone.

3.4. Double pipe heat exchangers

Chun et al. [103] experimentally analyzed convective heat transfer coefficient of nanofluids (alumina nanoparticles and transformer oil) through a double pipe heat exchanger system under laminar flow regime. The experimental results showed that the addition of nanoparticles in the fluid increases the average heat transfer coefficient of the system in laminar flow (Fig. 12). The surface properties of nanoparticles, particle loading, and particle shape are key factors for enhancing the heat transfer properties of nanofluids. These increases of heat transfer coefficients may be caused by the high concentration of nanoparticles in the wall side by the particle migration.

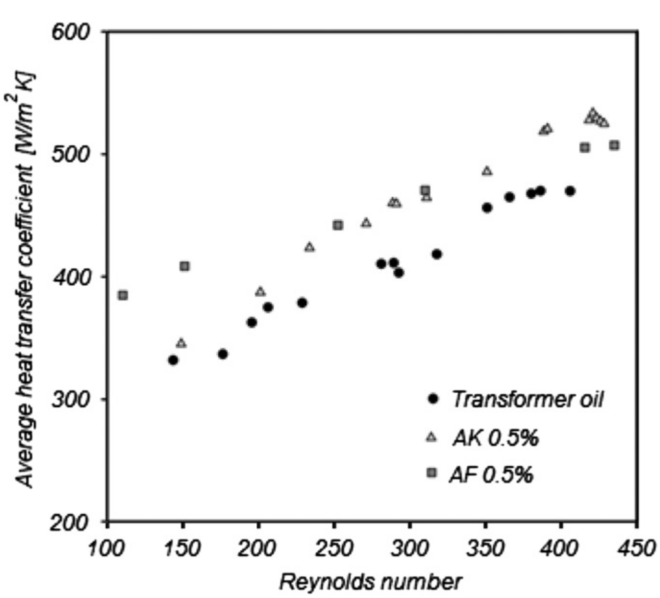

Fig. 12. Variation of heat transfer coefficient versus Reynolds number caused by the shape of nanoparticles (Chun et al. [103]).

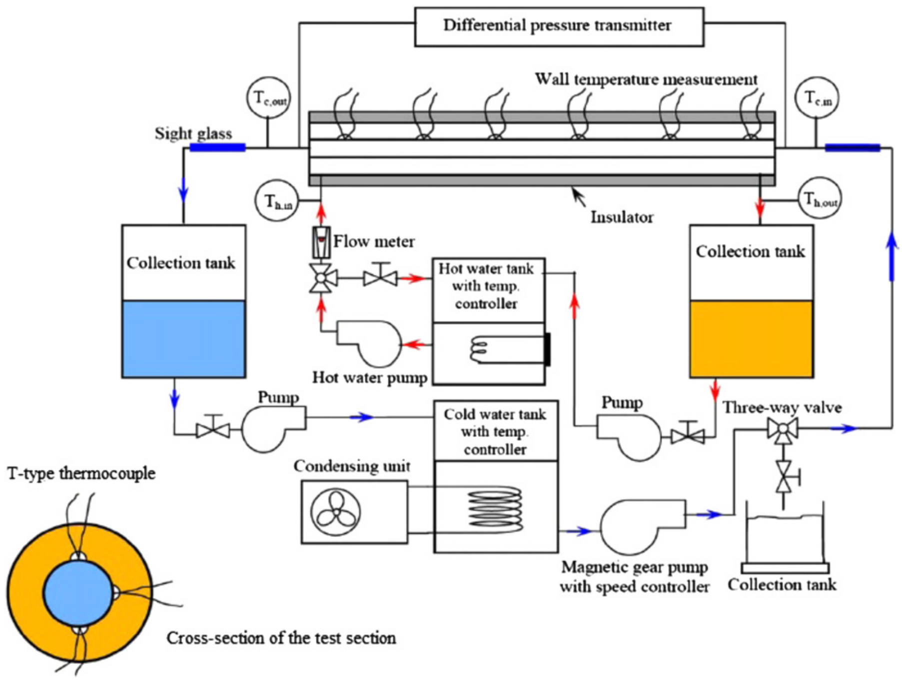

Fig. 13. Experimental setup for the study of the flow and heat transfer characteristics of the TiO_2/water nanofluid flow inside horizontal double-tube heat exchanger under turbulent flow (Duangthongsuk and Wongwises [80]).

Duangthongsuk and Wongwises [80] investigated the effect of thermophysical properties models on prediction of the heat transfer coefficient and also reported the heat transfer performance and friction characteristics of nanofluid under turbulent flow conditions. The 0.2 vol% TiO_2 naoparticles are used to disperse in the water.

The experimental apparatus (Fig. 13) used in this experiment consists of a test section, two receiver tanks, a magnetic gear pump, a hot water pump, a cooler tank, a hot water tank and a collection tank. The test section is a 1.5 m long counter flow horizontal double-tube heat exchanger with nanofluid flowing inside the tube while hot water flows in the annular. The inner tube is made from smooth copper tubing with a 9.53 mm outer diameter and an 8.13 mm inner diameter, while the outer tube is made from PVC tubing and has a 33.9 mm outer diameter and a 27.8 mm inner diameter. The inlet and exit temperatures of hot water are measured using T-type thermocouples which are inserted into the flow directly. The receiver tanks of 60 l are made from stainless steel to store the nanofluid and hot water leaving the test section. The cooler tank with a 4.2 kW cooling capacity and a thermostat is used to keep the nanofluid temperature constant. Similar to the cooler tank, a 3 kW electric heater with a thermostat was installed to keep the temperature of the hot water constant. The experiments were performed within the following ranges: the Reynolds number of the nanofluid varies in the approximate range of 3000–18,000, the temperature of the nanofluid is 15 °C, 20 °C and 25 °C, the mass flow rates of the hot water are 3 lpm and 4.5 lpm, the temperature of the hot water is 35 °C, 40 °C, 45 °C and 50 °C.

The results showed that the various thermophysical models have no significant effect on the predicted values of Nusselt number of the nanofluid. The results also indicated that the heat transfer coefficient of nanofluid is slightly greater than that of water by approximately 6–11%. The heat transfer coefficient of the nanofluid increases with an increase in the mass flow rate of the hot water and nanofluid, and increases with a decrease in the nanofluid temperature, and the temperature of the heating fluid has no significant effect on the heat transfer coefficient of the nanofluid. Finally, the use of the nanofluid has little penalty in pressure drop.

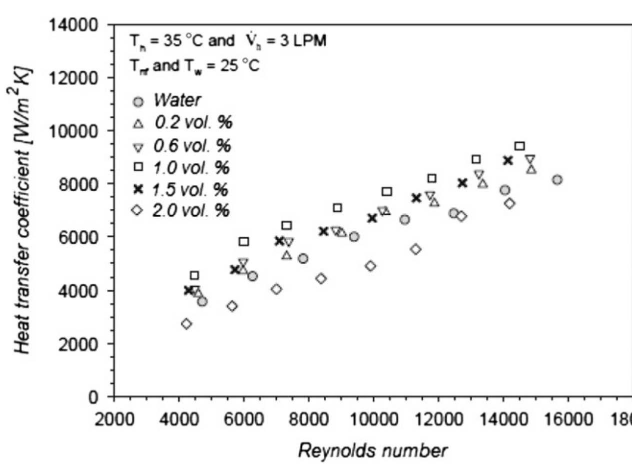

Fig. 14. Experimental heat transfer coefficient and Nusselt number for water and TiO_2/water nanofluids versus Reynolds number at various volume concentration (Duangthongsuk and Wongwises [60]).

Duangthongsuk and Wongwises [60] experimentally investigated the forced convective heat transfer and flow characteristics of a nanofluid consisting of water and 0.2–2 vol% TiO_2 nanoparticles flowing in a horizontal double-tube counter flow heat exchanger under turbulent flow conditions. The results showed that the heat transfer coefficient of nanofluid is higher than that of the base liquid and increased with increasing the Reynolds number and particle concentrations (Fig. 14). The heat transfer coefficient of nanofluids was approximately 26% greater than that of pure vol%, and the results also show that the heat transfer coefficient of the nanofluids at a volume concentration of 2.0 vol% was approximately 14% lower than that of base fluids for given conditions. For the pressure drop, the results show that the pressure drop of nanofluids was slightly higher than the base fluid and increases with increasing the volume concentrations.

Forced convection flows of nanofluids consisting of water with TiO_2 and Al_2O_3 nanoparticles in a horizontal tube with constant wall temperature have been investigated numerically by Demir et al. [104]. Numerical results have clearly showed that the use of nanofluids can significantly increase heat transfer capabilities of nanofluids even for relatively small particle volume fractions. Nanofluids with higher volume concentration have higher heat transfer enhancement and also have higher pressure drop. Therefore, judicious decision should be taken when selecting a nanofluid that will balance the heat transfer enhancement and the pressure drop penalty.

Zamzamian et al. [105] investigated forced convective heat transfer coefficient in turbulent flow in a double-pipe and plate heat exchangers with Al_2O_3 nanoparticles and CuO nanoparticles in ethylene glycol. The inner pipe of the double pipe heat exchanger was made of copper, 12 mm in diameter and 1 mm in thickness, with a heat exchange length of 70 cm. The shell was made of green pipes, 50.8 mm in diameter. The plate heat exchanger has 40 cm in height and 60 cm in length. The experiments were performed within the following conditions: the nanofluids temperatures were 45 °C, 60 °C, and 75 °C, volumeter flow rate was 3 l/min for nanofluids inside both heat exchangers and and 2.5 l/min for the cold water flowing inside the shell of the double pipe exchanger. The experimental system used in this study was shown schematically in Fig. 15.

The results showed that homogenously dispersed and stabilized nanoparticles enhance the forced convective heat transfer coefficient of the base fluid significantly. The greatest and smallest increases in our experiments were 49% and 3%, respectively. An evaluation of the effects of temperature and nanoparticle concentration revealed that with increasing temperature and nanoparticle concentration, a greater increase is observed in forced convective heat transfer coefficient. On the other hand, theoretical and experimental findings show a considerable discrepancy at higher temperatures and nanoparticle concentrations which may be accounted for by the fact that theoretical equations do not take temperature effects, nanoparticles fouling, nanofluid stabilizing methods, and type of stabilizing agent into account and they are not applicable to all nanofluids in all heat transfer procedures.

The heat transfer characteristics of double-tube helical heat exchangers using nanofluids (CuO and TiO_2 nanoparticles and water) under laminar flow conditions have been investigated numerically by Huminic [106]. The results showed that for 2% CuO nanoparticles in water and same mass flow rate in inner tube and annulus, the heat transfer rate of the nanofluid was approximately 14% greater than of pure water and the heat transfer rate of water from annulus than through the inner tube flowing nanofluids was approximately 19% greater than for the case which through the inner and outer tubes flow water. The numerical results also showed that the convective heat transfer coefficients of the nanofluids and water increased with increasing of the mass flow rate and with the Dean number (Fig. 16).

The convective heat transfer coefficient of silver–water nanofluids was investigated experimentally in a 4.3 mm inside diameter tube-in-tube counter-current heat transfer test section by Godson Asirvatham et al. [63]. Experimental results show that the suspended nanoparticles remarkably increase the heat transfer performance of the base fluid, water, under the same Reynolds number. The addition of 0.9 vol% silver particles in water enhances the heat transfer coefficients by 69.3%. Considering the factors affecting convective heat transfer characteristics of nanofluids, such as flow velocity, transport properties, volume fraction of nanoparticles, a new convective heat transfer correlation for nanofluids suspending metal nanoparticles has been developed. Comparison between experimental data and calculated results indicates that the correlation could take into account the main factors that affect heat transfer of the nanofluid and could be used to predict heat transfer coefficient of nanofluids with ±10% deviation.

4. Conclusions

Recently important theoretical and experimental research works on convective heat transfer appeared in the open literatures on the enhancement of heat transfer using suspensions of nanometer-sized solid particle materials, metallic or nonmetallic in base heat transfer fluids. Thus, this paper presents an overview of the recent investigations in the study the thermophysical characteristics of nanofluids and their role in heat transfer enhancement from heat exchangers. General correlations for the effective thermal conductivity, viscosity and Nusselt number of nanofluids are presented. Compared to the reported studies on thermal conductivity, investigations on convective heat transfer of nanofluids are limited. Most of the experimental and numerical studies showed that nanofluids exhibit an enhanced heat transfer coefficient compared to its base fluid and it increases significantly with increasing concentration of nanoparticles as well as Reynolds number.

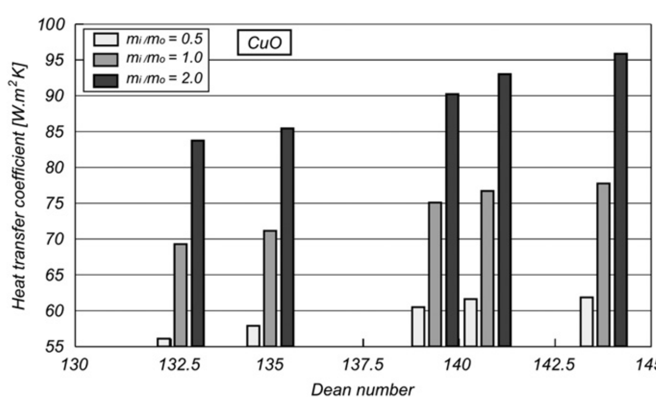

Fig. 16. Heat transfer coefficient for nanofluids versus Dean number at different volume concentration level (Huminic [106]).

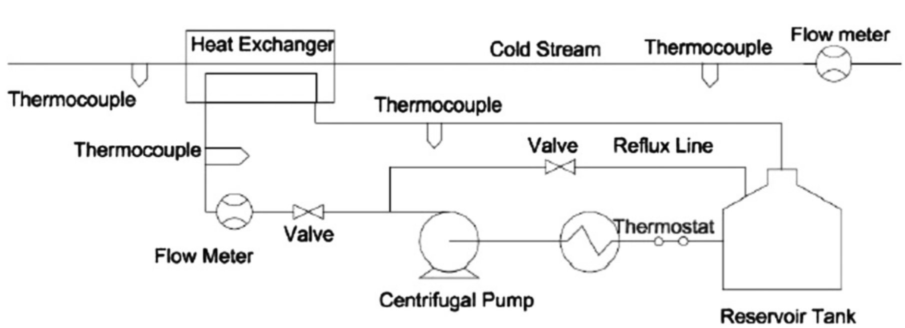

Fig. 15. Experimental setup for the study of the convective heat transfer in a double-pipe and plate heat exchangers with Al_2O_3/EG and CuO/EG nanofluids under turbulent flow (Zamzamian et al. [105]).

The enhancement of the heat transfer capability of nanofluids makes their use in heat exchangers an interesting option, leading to better system performance and the resulting advantage in energy efficiency. On the other side, nanofluids stability and its production cost are major factors that hinder the commercialization of nanofluids.

Further theoretical and experimental research investigations on the effective thermal conductivity and viscosity are needed to demonstrate the potential of nanofluids and to understand the heat transfer characteristics of nanofluids as well as to identify new and unique applications for these fields.

Acknowledgements

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